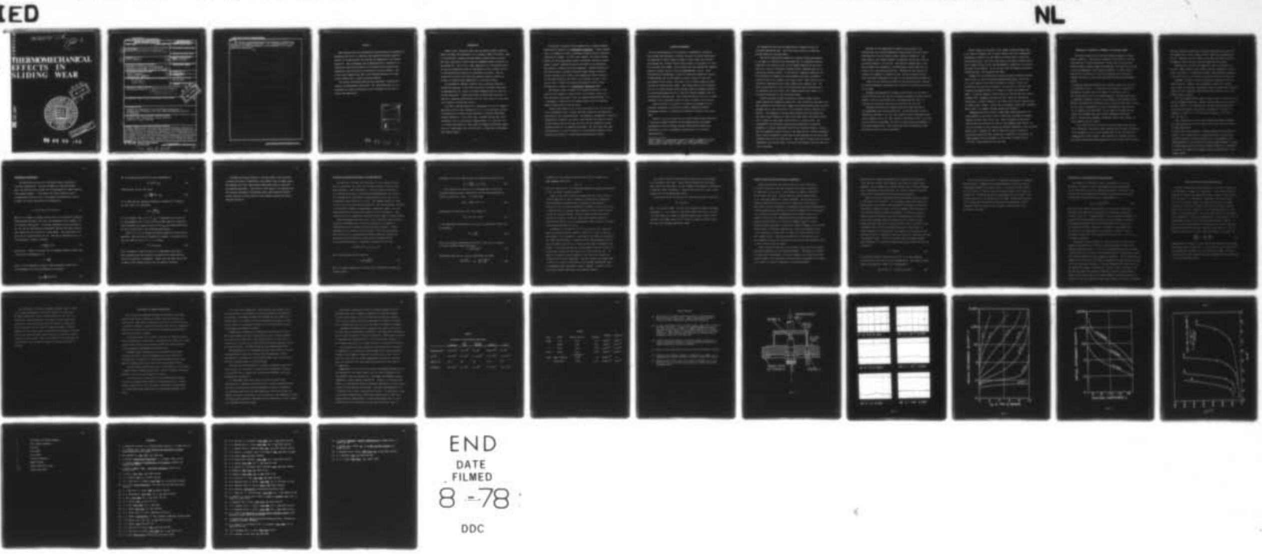
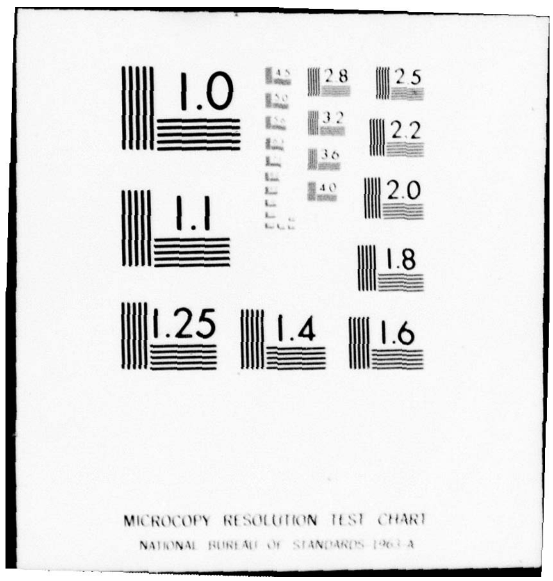
AD-A055 318   NORTHWESTERN UNIV EVANSTON ILL DEPT OF MECHANICAL E--ETC   F/G 20/11
THERMOMECHANICAL EFFECTS IN SLIDING WEAR,(U)
APR 78 R A BURTON   N00014-75-C-0761

UNCLASSIFIED   NL

1 OF 1
AD
A055 318



END
DATE
FILMED
8-78
DDC


1.0
1.1
1.25
1.4
1.6
1.8
2.0
2.2
2.5
2.8
3.2
3.6
4.0
MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

THERMOMECHANICAL EFFECTS IN SLIDING WEAR

NORTHWESTERN UNIVERSITY
QUAECUMQUE · SUNT · VERA
ὁ λόγος πλήρης χάριτος καὶ ἀληθείας
1851

Unclassified

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) Thermomechanical Effects in Sliding Wear		5. TYPE OF REPORT & PERIOD COVERED
		6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) Ralph A. Burton		8. CONTRACT OR GRANT NUMBER(s) N00014-75-C-0761
9. PERFORMING ORGANIZATION NAME AND ADDRESS Department of Mechanical Engineering and Astronautical Sciences, The Technological Inst., Northwestern University, Evanston, IL 60201		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS Keith Ellingsworth, Office of Naval Research, Arlington, VA 22217		12. REPORT DATE April 1978
		13. NUMBER OF PAGES 43
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)		15. SECURITY CLASS. (of this report) Unclassified
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

6 10 15 11 12 46p.

16. DISTRIBUTION STATEMENT (of this Report)

Distribution Unlimited

DISTRIBUTION STATEMENT A

Approved for public release; Distribution Unlimited

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

Prepared for presentation at the International Conference on Fundamentals of Tribology, June 19-27, 1978, Massachusetts Institute of Technology.

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Friction, wear, thermoelastic, surface mechanics, lubrication, seals, tribology

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

When sliding occurs with significant frictional heating, thermoelastic deformation may lead to a transition from smoothly distributed asperity contact to a condition where the surfaces are supported by a few thermal asperities. This circumstance may be associated with a transition to a condition of severe wear because of the elevated contact pressure and temperature, and also because of production of tensile stresses. This second stress component may lead to heat checking whereupon the rough checked surface acts to abrade the mating material.

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE
S/N 0102-014-6601

Unlcassified

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

403 502

LB

SECURITY CLASSIFICATION OF THIS PAGE(When Data Entered)

20.

The factors influencing transition are discussed, including wear, cooling, and hydrodynamic lubrication. The transitioned state is also discussed as to stress distribution, rate of movement of the contact patches and temperatures.

SECURITY CLASSIFICATION OF THIS PAGE(When Data Entered)

Abstract

When sliding occurs with significant frictional heating, thermoelastic deformation may lead to a transition from smoothly distributed asperity contact to a condition where the surfaces are supported by a few thermal asperities. This circumstance may be associated with a transition to a condition of severe wear because of the elevated contact pressure and temperature, and also because of production of tensile stresses. This second stress component may lead to heat checking whereupon the rough checked surface acts to abrade the mating material.

The factors influencing transition are discussed, including wear, cooling, and hydrodynamic lubrication. The transitioned state is also discussed as to stress distribution, rate of movement of the contact patches and temperatures.

INTRODUCTION

Modern contact theory has drawn upon the powerful ideas of Holm (1), Bowden and Tabor (2) and Merchant (3) to provide a model of surfaces, rough on the microscale, forming localized highly stressed contact junctions. This model replaced the earlier asperity-engagement models of Belidor (4) and Coulomb (5). The idea of a high temperature in a frictionally heated contact was introduced by Blok (6) and applied by him to explain failure of lubricated Hertzian contacts as encountered in gearing. Holm (7), Archard (8) and others have drawn upon similar concepts to provide an estimate of localized high temperatures in frictional sliding.

As experimental findings accumulate and as analytical modeling is extended and improved, it has become clear that the asperity model must be supplemented by the concept of thermal deformation and the creation of "thermal asperities". This concept helps to explain many instances of high wear, anomalous partitioning of heat (9), and a chain of events sometimes leading to catastrophic failure.

The idea of thermal asperities is supplemental to that of roughness asperities, and in no way antagonistic to it. On the one hand a process of instability can produce thermal asperities on surfaces approaching absolute smoothness. On the other hand, a thermal asperity may itself correspond to one of the original roughness peaks; or it may give rise to a contact patch wherein a large number of roughness asperities engage those of a second body, just as they would in a large area of undeformed flat surface contact.

The process of transition from nominally flat to highly deformed surfaces may be spoken of as thermoelastic transition. It may sometimes occur in a sequence of stable, continuously related states as operating conditions are changed. At other times, however, the stably evolving behavior of the sliding system crosses a threshold, whereupon a sudden change of contact conditions occurs as the result of an instability. This involves a feedback loop which comprises: localized elevation of frictional heating, resultant localized thermal bulging, localized pressure increase as the result of the bulging and further elevation of frictional heating as the result of the pressure increase. This process, when it leads to an accelerated change of contact stress distribution, is spoken of as thermoelastic instability (TEI).

The ultimate result of growth of the thermal disturbance is the parting of the surfaces in some portions of the nominal contact area, with gap height being several times the roughness-asperity height. As a consequence of reduction of the nominal contact area the remaining contact patches acquire elevated stress.

Contact patch formation can occur in lubricated as well as dry contact and is influenced by wear, cooling, materials properties, and macroscopic constraints on the contacting bodies. The physics of thermoelastic transition, thermoelastic instability and contact patch behavior have begun to receive serious interest only in the past few years, and the explanation of many important effects is not generally well known. For this reason space must be devoted here to a brief introduction of the phenomena involved, before discussion may proceed to problem areas needing further research.

HISTORICAL BACKGROUND

Prior to the introduction of the concepts of thermoelastic transition Ling and Mow (10) developed an influence function for surface displacement for high-Peclet* number sliding of bodies in plane strain, and outlined the procedure for treating a moving contact patch on the surface of a slab, also for high Peclet number. Mow and Cheng (11) have examined the companion problem of thermal stress in plane elastohydrodynamic contact. Early investigations of thermoelastic effects on lubricated sliding were reported by Korovchinsky (12) for the change of contact stress of a ball with frictionally heated contact, and by Nica (13) for radius change of journal bearings with fixed external radius. The bearing work has been broadened (14, 15) to include cooling effects and more realistic boundary conditions for fluid-film bearings, and also to show that similar phenomena exist for rolling contact bearings (16). Most interesting here is the role of thermal expansion in the catastrophic chain of events of seizure of the bearing.

Sibley and Allen (17) carried out a series of experiments on seal materials, developing a criterion for thermal checking and showing photographic evidence of systematically moving hot patches in the contact zone.

Interest in contact instabilities and patch formation was accelerated by the work of Barber (18,19,20) who has demonstrated the phenomenon experimentally and has provided analyses which partially explain his observations, as well as fundamental contributions to the field of thermoelasticity.

---

*Peclet number is a dimensionless measure of speed of movement of a heat source, being of the form $c\ell/k$, where c is speed of movement, ℓ is a characteristic length (patch length) and k is thermal diffusivity.

His explanations draw upon the modification of asperity contact by frictional heating and wear. His interest was initially in explaining hot-spot effects in railroad brakes.

Dow (21,22) addressed the problem of a scraper sliding perpendicular to its edge on a conductive slab and showed that (1) instability would be predicted in the absence of wear and (2) would be modified by wear which would: (a) raise the instability threshold and (b) sometimes give rise to oscillating constant pressure. He carried out a numerical simulation (23) which predicts the formation of contact patches, and their translation along the edge of the scraper as the result of wear. More recently he has carried out experiments which vividly display patch formation (24). The onset of such patches is close to his theoretical critical sliding speed for instability for dry contact; but although qualitatively similar, it is only poorly predicted for wick lubricated contact.

Kennedy and Ling attacked the problem of severely loaded aircraft brakes drawing upon numerical analysis (25), later followed by experiments (26). They postulated a wear model which suggests the mild-wear/severe-wear transition, assuming that no wear occurs until a critical shear stress is reached in the material. For sandwiches of moving disks alternated with stationary ones, they find contact reducing to a band at the outer radius, then moving inward, only to repeat the sweep again several times in a typical "stop". The confinement of the disk brake causes material properties and operating parameters to interact differently than for the "floating" scraper. Furthermore the appearance of patch contact in the brake may be a transient phenomenon rather than an instability in that there is some evidence that the sweep would ultimately die out as the brake wore-in. This cannot be tested since the overall temperature rises severely even in the brief runs reported, and thus limits the time of operation.

Nerlikar (27) has addressed the problem of ring contact in an idealized face seal sliding tangentially, and has shown that the relative conductivities of the two bodies strongly influence instability. The least stable is the thermal conductor on insulator; and the most stable is a material sliding on its own kind. This has been extended (28) to show that extremely thin contaminant, solid-lubricant or oxide films can play a major role in determining instability. Nerlikar has also explored the influence of roughness on instability (29), and has calculated the gap width and contact temperatures for seals in the contact-patch configuration.

Lebeck (30) has improved the model for sealing-ring contact, allowing for thin-beam bending, and showing how to account for cooling from the sides of the rings.

Kilaparti has addressed the problem of patch contact in the absence of wear (31) and with wear (32,33) developing an improved influence function similar to that of Ling and Mow (34), and discovering that there is a critical wear coefficient above which instability will not occur.

Banerjee has treated the idealized seal with liquid lubricant in the hydrodynamic short-bearing regime (35). He has predicted a range of instability, modified at the thin film extreme by elastic deformation under contact pressure, and at the thick extreme by convection of heat in the film. He has carried out experiments (See fig. 1,2) which show the instability to occur where expected (36). More recent work (37) has modified his model but does not alter the basic conclusions.

Closely related to this work is that of Hahn and Kettleborough (38), Ettles (39) and Tanaguchi (40) on thrust washers, where only thermoelastic deformation can explain the load support of notched rings in lubricated, wide-bearing contact. Why these rings (and for that matter sliding systems such as engine pistons) do not show thermoelastic instability is a question that must yet be answered. We can only note here that Banerjee predicted a stabilizing effect from increased face width.

Heckmann (41) has reassessed the ring and scraper instability problems from the point of view of controlling-dimensionless-groups, there being w/w^* a wear measure, and H and ζ two groups for cooling. He has also extended the seal and scraper problem to line contact on a slab (42) thus bringing three-dimensionality in to replace the earlier two-dimensional models. An axially symmetric contact patch on a slab has also been treated elsewhere (43). Heckmann's studies show that, except for quantitative differences, a cylinder sliding in line contact with a slab is substantially the same as Dow's scraper. Indeed, Dow's experimental pieces (24), simulating Wankel seals, lie somewhere between scrapers and cylinders.

Wangkrajong (44) has carried out high speed sliding experiments for carbon-graphite on mild steel, where at 400 ft/sec and higher the evidence suggests extremely small contact area. Work in progress on high speed turbine blade and labyrinth seal rubs not only suggests patch contact, but calls for analyses to treat bouncing and vibratory compliance of the contacting members (45). Although this 1000 to 1500 ft/sec contact represents a spectacular application where thermoelastic instability is inevitable, one should remember that the phenomenon is also found at speeds as low as 10 ft/sec in some geometries (see also (46)).

THERMOELASTIC TRANSITION OF NOMINALLY FLAT SLIDING CONTACT

Face-seals represent one of the best definable examples of thermoelastic transition. Such seals are ordinarily in the form of concentric rings meeting at a plane perpendicular to their axis, with contacting surfaces lapped to quarter-light-wave smoothness, and supported by bellows or O-ring arrangements to make them self aligning. At times they may operate with no lubricant, or they may incorporate solid lubricants, boundary lubricants or even liquid lubricants which provide hydrodynamic support.

An experiment (36) which simulates the frictional contact of a face seal is illustrated in fig. 1. A cylindrical cup of metal is inverted and supported by an axial stem in the chuck of a drill press (fig. 1). The circular edge of the cup is pressed against a flat glass plate which is supported in gimbals to permit a self aligning, almost-uniform contact. Whether the surfaces are lubricated or dry, as the cup turns about its axis, one will find a temperature rise which is determined by speed and load but is typically only about 100^{o}C. As the speed is increased, one may observe the interface from below, looking through the glass plate; and at a characteristic, somewhat reproducible speed he will observe red light flashing from the contact.

If the glass and cup are inverted, so that the glass turns and the cup is stationary, the event of light evolution will be found to correspond to the appearance of two or three discrete spots on the cup, which glow orange-red and twinkle slightly. When operation is continued the spots move slowly from their original position at a rate of about 10^{o} per minute.

Thus even though the glass may be turning at 2000 rpm the hot spots will take roughly half an hour to make a circuit around the circumference of the cup. Typically they will remain as two spots 180^o from one another, although for short times more spots may appear and later vanish.

Examination of the metal cup immediately after the formation of a hot spot shows evidence of some plastic flow and a darkened or "burned" patch on the metal. If examination follows a longer period of operation, it will show tracks on the surface over which the spots have traversed.

Although the spots are red-hot the general temperature of the cup remains moderate. Damage is not catastrophic because only a shallow layer of metal is affected. Nevertheless there is evidence that prolonged operation in this regime may significantly affect the wear rate and can produce surface cracking.

The first explanation that comes to mind is that the contact is at the peaks of roughness asperities. However, when one measures the contour of the metal surface in the deformed state one finds the hot spots may be associated with pimples on the surface, as high as 10^{-3} cm while the roughness is about 2.5×10^{-5} cm and initial waviness is 10^{-4} cm or less (see fig. 2).

More interesting is the fact that if speed of sliding is reduced, the pimples shrink back into the surface and in some cases are replaced by slight depressions.

To explain this we may postulate a thermal expansion which produces the pimples. Then, because there is concentrated contact on their peaks, frictional heating is concentrated there so as to maintain the local thermal expansion. Since frictional heating is speed dependent, it would be reduced upon reduction of speed, and would ultimately be insufficient to maintain the height of the pimples, hence the collapse back to flat surface contact.

Conditions for Self-Sustaining Patch-Contact

To show the factors which permit the existence of patch contact, let us assume that the edge of a plate rubs against a rigid thermal insulator, and that the contact patch is 2ℓ wide on the edge of the plate. Not, at this point, knowing the contact pressure distribution we assume it to be the uniform $\bar{p}$; thus if the sliding speed is V and friction coefficient is μ, the heat input per unit of area is

$$q = \mu \bar{p} V \tag{1}$$

It has been shown that the curvature of the edge of a plate is related to heat input such that

$$\frac{d^2 v}{dx^2} = \frac{1}{R} = \frac{\alpha q}{K} \tag{2}$$

where R is radius of curvature, v is normal displacement and K is thermal conductivity. At this point we may ask what contact pressure would be required to flatten this curvature across the contact zone, and turn to the well known relationship for Hertzian contact

$$\ell/R = 2.3\ \bar{p}/E^* \tag{3}$$

Eliminating R between Eq. (2) and Eq. (3) and substituting for q from Eq. (1)

$$\ell = \frac{2.3K}{E^* V \mu \alpha} \tag{4}$$

More exact studies bear out the essential correctness of this finding, which shows two interesting features:

(1) the sliding speed required to sustain a patch of a particular ℓ is independent of the loading.

(2) there is nothing to prevent there being one, three or any number of contact patches on a given sliding edge, thus posing a problem of uniqueness.

Commenting on the first, we note that a load influence could appear in the physical problem if μ, E, α, K depend on temperature or stress level. Commenting on the second, there is a problem of uniqueness which disappears when a second consideration is introduced, this being the stability of the solution under small disturbances. A test for this can easily be introduced at the present level of argument.

Nerlikar's work (29) showed that if the contact load on a patch is increased the surfaces will move further apart in the region between patches. This paradoxical effect causes leakage to increase with increased face load on a seal. It also gives rise to an instability which causes the number of contact patches to reduce to the minimum number consistent with mechanical equilibrium. For a self aligning seal the minimum is two patches; for one where tilting is constrained one patch is the minimum.

To visualize this we note that the seal can be developed into an infinite plate, where contact patches are cyclically repeated. If alternate patches are reduced in load the others must experience load increase. These load shifts give rise to displacements which further increment the load changes until ultimately the patches with decreasing load break contact.

Thermoelastic Instability

The question arises now as to how smooth contact transitions to the patch configuration. This may be posed as a stability problem, where the conditions may be found for run away growth of a small departure from uniform pressure. If the edge of the cup is allowed to have a circumferential waviness, and if the waviness is pressed flat by axial loading, the contact pressure may be represented by

$$p = \bar{p} + \hat{p} \cos \theta = \bar{p} + \hat{p} \cos \kappa x \qquad (5)$$

Where θ is a measure of angular position and x is a curvilinear coordinate running around the edge of the cup. The pressure $\hat{p}$, will be spoken of as the "pressure perturbation". Frictional heating will occur according to eq. (1), but the uniform portion produced by $\mu\bar{p}V$ will not affect waviness, only causing the cup to flare out a small amount. The perturbation will give rise to a distribution: $\hat{q} \cos \kappa x$. This may be inserted into eq. (2) and integrated to produce a waviness:

$$v = \frac{\alpha}{K} \frac{q}{\kappa^2} \cos \kappa x \qquad (6)$$

From elastic theory it is known that the pressure required to press flat a sine wave of displacement v is

$$p = \frac{E\kappa v}{2} \qquad (7)$$

Hence, if the displacement increment from the heating is held flat at will supplement the initial perturbation by an amount

$$p_{th} = \frac{E\alpha}{2K\kappa} \hat{q} \cos \kappa x' \qquad (8)$$

But the heating perturbation will now be determined by

$$\hat{q} = \mu V(\hat{p} + p_{th}) \tag{9}$$

Inserting this into eq. (8) yields

$$p_{th} = \frac{E\alpha\mu V}{2K\kappa} (\hat{p} + p_{th}) \tag{10}$$

Let us draw upon the convenient grouping of variables V^* to simplify eq. (10); where $V^* = (2K\kappa)/(E\alpha\mu)$

$$p_{th} = \frac{V\hat{\beta}}{(V - V^*)} \tag{11}$$

It is now apparent that if $V = V^*$, $p_{th} \to \infty$ irrespective of the size of $\hat{\beta}$. This suggests that V^* is the critical sliding speed for transition; since ultimately the negative portion of the p_{th} wave would exceed $\bar{p}$ and the surfaces would part, thus initiating patch contact.

Table 1 is a list of thermal and elastic properties of some typical materials. When the material aluminum, for example, is selected for the cup, one finds that for $\mu = 0.1$, $\kappa = 2$ rad/cm.

$$V^* = 0.64 \text{ m/sec} \tag{12}$$

This corresponds to about 150 rpm for a 75 mm diameter cup; yet no major transition has been observed in actuality up to about 2500 rpm. Clearly the analysis is incomplete. Indeed, wear and other factors will be shown to have dominant effects upon the growth of waviness.

Although this analysis predicts a critical speed at which waviness is greatly multiplied in amplitude it also predicts that at higher speeds the amplitude will drop; and extremely high speeds would be associated with improved smoothness. This would at first appear to be something like the resonance phenomenon of synchronous whirl, where machines reqularly operate above the critical speed; but more careful analysis will show an important difference.

Conditions for Exponential Growth of Surface Waviness

The existence of steady state solutions as in the previous section does not assure that the system will be bound to these if they are themselves unstable to small disturbances. A test for such an instability of a solution may be made in several well established ways, one of which is to hypothesize an exponentially growing waviness and to test for the circumstances under which this can exist. The argument behind this would be that if (1) the system were disturbed and if (2) the disturbance contained a time dependent component corresponding to exponential growth, then such growth would continue no matter how small the initial disturbance. Thus a grain of dirt or a temperature fluctuation could trigger the growth process.

Before proceeding with such a test, let us review the initial analysis, noting that sinusoidal waves of contact pressure were accompanied by sinusoidal waves of frictional heating and thermal expansion. Further examination of the equation for heat transfer in the tube would suggest also that there is a corresponding sinusoidal temperature perturbation, proportional to $\hat{p}$ and having its peak where $\hat{p}$ is maximal (22,35). Let us therefore postulate a temperature perturbation of the type

$$T = \hat{T}e^{-by}\, e^{i\beta t} \cos \kappa x = T_s\, e^{-by} \qquad (13)$$

Then contact pressure will be given by

$$p = E\alpha T_s \left(\frac{+\kappa}{\kappa + b}\right) \qquad (14)$$

Here is it again assumed that the axial load is sufficient to assure full surface contact.

Returning to eq. (13) we find the heat flux through the surface will be

$$q = -K\left(\frac{\partial T}{\partial y}\right)_{y=0} = K\,bT_s \qquad (15)$$

Let us require now, that for a self sustained wave to exist the frictional heating μVp must correspond to the heat passing through the surface as given in eq. (11) . It follows that

$$KbT_s = \mu VE\alpha T_s\,(\kappa/b + \kappa) \qquad (16)$$

Drawing upon the definition of V^*, this reduces to

$$(1 + b/\kappa)\; b/\kappa = 2V/V^* \qquad (17)$$

If the temperature distribution in eq. (13) is to satisfy the Fourier heat flow equation,

$$\nabla^2 T = \frac{1}{\kappa}\frac{\partial T}{\partial t} \qquad (18)$$

then, the following relationship must prevail, when b must be positive to satisfy distant boundary conditions.

$$b = \sqrt{\kappa^2 + i\beta/\kappa} \qquad (19)$$

Substituting this into eq. (17) and simplifying, one finds

$$\sqrt{1 + \beta/\kappa^2} = -\,1/2 \pm \sqrt{\frac{1}{4} + 2\,\frac{V}{V^*}} \qquad (20)$$

Examination of this equation will show that $i\beta$ may be replaced by a <u>real</u>, <u>positive</u> number when

$$V/V^* \geq 1$$

Hence the condition for a self sustained exponentially growing perturbation is that the speed of sliding exceed V^*.

Again we note that the rational behind this test is that if such a condition <u>can</u> exist it may be triggered by fluctuations in operating conditions during extended periods of nominally steady operation. Obviously if such a pressure perturbation continues to grow it will reach a condition where the negative lobes exceed the initial contact pressure provided by the axial loading. This will lead to regions of parting between the surfaces. As has been shown this condition ultimately stabilizes the contact in a new configuration, where contact is in patches spaced around the edges of the cup.

Returning to the definition of V^* it is seen that the transition speed is dependent upon wave number κ, which is π/λ, and λ is half the wave length of the perturbation. Since small κ (or large wave length) has the lowest critical speed, it follows that upon advancing speed up from zero, instability will first be reached for the longest permissible wave length. If the glass and cup are stiffly mounted to prevent tilting, the critical wave length will correspond to one cycle around the cup. If, however, the glass is held in gimbals it will tilt so as to make the existence of $p = \hat{p} \cos \theta$ impossible. The next longest wave length would be that for $p = \hat{p} \cos 2\theta$ and would have two pressure maximum 180° apart. In experiments such a disturbance tends to dominate, in that hot spots (or contact patches) 180° apart are frequently observed.

As a word of caution, these derivations apply to the cup described, or to relatively stiff rings. Lebeck (30) has pointed out that slender rings, which obey beam theory, are more compliant and provide a stabilization of long wave length disturbances. They may also give rise to more contact patches than the minimum mentioned above.

Returning to Eq. 11 the equation for critical speed may be rewritten as

$$V^* = 4\pi K/E\alpha\mu\lambda$$

where λ is the wave length. For very thin bearings the longest wavelength permissible is 3.7 h, where h is the axial length of the ring. As speed is raised longer wave lengths or spacings between contact patches will be tolerated, the length rising as $V^{1/3}$, until the number of patches reaches the above stated minimum permissible number.

Effects of Heat Conduction and Wear on Instability

Because uniform heat flow down the axis of a ring or tube does not lead to waviness in the axial direction, flat contact instability is independent of this factor. Patch contact is, however, influenced by the overall heat flow. In both cases the relative conductivities of the contacting bodies are of major importance. For the tube-on-tube geometry, Nerlikar (27) has carried out a theoretical study allowing both bodies to have the properties of aluminum, with one having hypothetically altered conductivity K_h (the effect of this on diffusivity is also accounted for). Results for critical sliding speed are as shown in Fig. 3, as a function of the ratio of conductivities and the friction coefficient. Except for high friction, the curves turn upward at $K_h/K = 1$, and the critical sliding speed goes to infinity for material sliding on its own kind. Another effect of differing conductivities is that the perturbation wave is not stationary in either body, but moves more slowly relative to the more conductive body. Even when both bodies are of the same material, the asymmetry necessary for instability can arise from thin insulating films on one or either of the two bodies.

Such a film will offer much less resistance to heat flow for low-speed disturbance movement than for high-speed movement. Consequently a condition can arise where a disturbance is nearly stationary in one body and has high relative speed on the second body, and the apparent resistances of the bodies can be such as to lead to instability at low sliding speeds.

Calculations for aluminum-on-aluminum with a glassy film of thickness λ show that for infinite film thickness on one body the critical speed approaches that of conductor on insulator whereas film of reduced thickness lead to increased critical speed (Fig. 4). There is however no critical speed for zero film thickness or zero friction coefficient.

Figure 5 is based on calculations where frictional heating is assumed to take place at the interface. It is possible, however that some of the heating may be due to plastic deformation below the surface, and thus the film would not impede the flow of this heat component into the material. This question would bear further investigation. It does not, however obscure the fact that even if only half the generated heat is affected by the film, instability would be called for, yet it would be absent without the film. When wear is present (42,22) it may be treated in terms of Archard's wear law (47) in modified form, $dv/dt = wpV$, where p is contact pressure, V is sliding speed. The quantity w is an empirical wear coefficient which is influenced by material properties, temperature and other environmental factors. The wear coefficient may be nondimensionalized upon dividing by w^*, a group of variables which may be called critical wear coefficient.

$$w^* \equiv 2\alpha\mu k/K \tag{21}$$

For conductor sliding on insulator, when $w/w^* \geq 1$ in the conductor, instability will not occur for finite sliding speed. The critical sliding speed in the presence of wear, V_w^*, is approximately

$$V_w^* = V^* \left[1/(1 - 1.2\, w/w^*)\right] \exp\,(w/w^*) \tag{22}$$

Dow (22) has treated the case of a scraper running against a conductive drum, with wear. Heckmann (41) has generalized these results in terms of dimensionless quantities as shown in Fig. 5. The critical sliding speed is set in ratio to V^* for the scraper for several values of the conduction parameter ζ. The physical quantities incorporated into ζ are those of the scraper except for k, the diffusivity of the drum. Cooling from the sides of the scraper (and also circular seals) was also investigated; and for reasonable values of convection heat transfer coefficient the effects were found to be small.

Instability in a Hydrodynamically Lubricated Seal

The instability illustrated in Fig. 2 took place in a seal with sufficient oil present to assure hydrodynamic, short-bearing support for the faces, which were self aligning and supported a fixed axial load. Banerjee (35) has investigated the stability of seals with constant mean film thickness and found the critical sliding speed to be

$$U_{crit} = (K_m/\alpha_m \eta)^{1/2}\, \bar{h}\kappa \qquad (23)$$

where metal runs against rigid insulator as in the prior illustrations. Strikingly good experimental confirmation of this result was obtained as shown in Fig. (7). An improved model, where the constant $\bar{h}$ condition is relaxed, has been investigated and has led to the paradoxical conclusion that quasi-equilibrium states of the system approach, but do not cross the stability threshold. Because of the approach to the conditions of eq. (23) it is not surprising that the equation is satisfied. Recent experiments confirm that steady operation, with great care, can be carried out without transition.

Turning now to Dow's results for a scraper running on a drum, Table 2 compares the measured wear coefficient w and that required to predict the observed transition speed, w_c. Note that for combinations run dry, agreement is good except for the steel on steel which may have been influenced by overall heat transfer through the contact after the transition. For the lubricated runs prediction is poor but the transition was observed to be qualitatively the same as for the dry contact. Actually, it was the deformed patch-contact state that was observed as evidence of transition. In view of this it is possible that the friction coefficient and wear coefficient on the hot patches would be nearer to dry contact than for the smooth sliding of lubricated material. If that is so, then the similarity to dry contact behavior can easily be explained.

PATCH CONTACT WITH HEAT TRANSFER AND WEAR

Both the experiments of Dow and Banerjee, cited above, suggest that if the deformed state can exist it may exist. In the earlier studies of conductor on insulator the condition for patch contact to exist was shown to be the same as for thermoelastic instability. When more realistic models are considered, however, the formation of patches is strongly influenced by overall heat transfer from one body to another, whereas idealized flat surface instability is not, being dependent only on the growth of zero-average waves — at least for the case of the axisymmetric ring configuration. This can be demonstrated by referring back to Eq. (2) and adapting it to the case of heat flow from one body to another. If both are of the same material the one losing heat will curve inward and the one receiving heat will bulge outward the same amount. If they are different there will be a relative curvature change (see also Ref. 49)

$$\left(\frac{d^2V}{dx^2}\right)_{rel} = \left(\frac{\alpha_1}{K_1} - \frac{\alpha_2}{K_2}\right) q \tag{24}$$

Depending upon the sign of the heat flow and the materials, this influence may greatly alter the patch size. It may also give rise to patch contact for normally flat surfaces even in the absence of sliding. For these reasons, patch contact observations require knowledge of heat partitioning for their full interpretation.

Although study of the stationary patch with or without wear has provided considerable physical insight, it can easily be shown that the patches must be moving relative to both contacting surfaces when wear is present and each body is a thermal conductor. Even at low Peclet number the nature of the stress distribution is strongly affected by this movement. Kilaparti (32) has examined the case of the uniformly heated patch (q = constant), and has shown that, as the patch moves, wear causes a reduction of outward displacement v from leading edge to trailing edge. At the same time receipt of heat causes a rise in v over the same distance. Hence, if the patch is in contact with a rigid body these two effects must cancel one another. It follows that speed of traversal goes up with wear rate, and must exceed infinity for wear coefficient exceeding W^*. Even cases of so-called severe wear of several abradable materials lie below $W/W^* = 1$. A more detailed study (50) has shown that the pressure distribution on the contact becomes nearly triangular as wear rate is increased; but the study was limited to modestly high Peclet number because the numerical scheme required an excessively fine grid for higher values. A third study directed toward higher Peclet number (33) has reviewed the work of Ling and Mow (10) and suggested a simplified relationship for surface displacement:

$$\int_x^{\infty} \frac{2\alpha k}{cK} q(\xi)\,d\xi + \int_{-\infty}^{x} \frac{2}{\sqrt{\pi}} \left(\frac{k}{c}\right)^{3/2} \frac{\alpha}{K} \frac{q(\xi)\,d\xi}{\sqrt{x - \xi}} = \delta(x) \qquad (25)$$

Derivations drawing upon this function have supported the earlier prediction of nearly triangular contact pressure distribution.

Contact Stress

Recent calculations show that a tensile stress may appear at the edge of a moving contact patch with uniformly distributed heat input, this being given by

$$\sigma_{xth} = \frac{E\alpha Q}{K(1-\nu)\varepsilon} (0.8 \ Pe\ell) \tag{26}$$

where $Pe\ell = c\ell/k\pi$. In one example where the contact patch was virtually stationary in the carbon of a carbon/cast-iron contact, the rate of movement c relative to the iron would be approximately the sliding speed. For both bodies close to ambient temperature in the bulk, almost all of the frictional heat would be expected to pass into the iron, hence $Q \approx \mu PV$. Under these circumstances Eq. (26) becomes (for $\nu = 0.3$, and $\ell/\varepsilon = 1$)

$$\sigma_{xth} = \frac{1.14 \ E\alpha P}{Kk} \left(\frac{\ell}{\varepsilon}\right) V^2 \tag{27}$$

and for V in m/sec and σ_{xth} in bars one finds upon substituting the properties of cast iron

$$\sigma_{xth} = 26{,}394V^2$$

obviously this can lead to large stresses.

On the one hand this gives an easy explanation of surface heat checking where patch contact passes over the surface. At the same time it raises the question as to why heat checking is not observed more often. One possible explanation may be that the depth affected is shallow. It is possible also that the numerical magnitude may be reduced by substituting a different distribution of heat input on the patch. Whatever the modifying factors may be they offer an attractive problem for further investigation.

An illustration of such heat checking is shown in Fig. 6, which is an enlarged photograph of the band of contact of a large seal. The light colored band is over 2 mm wide and lies on the face of a hard brittle alloy which has been run against a carbon ring with a raised nose. The tiny hairline cracks run across the contact band and are believed to be the result of tangential tension. Evidence suggests that the thermal asperity forms and moves slowly on the carbon, and that the above described tension arises in the metal as it passes through the contact patch. There is no evidence of gross heating, and such heating is not necessary to produce the stresses necessary for cracking.

DIRECTIONS FOR FURTHER INVESTIGATION

One of the most troublesome problems left unsolved in the above reported work involves delineating the path from stable operation to patch contact. When this is understood we will be in a better position to avoid the formation of patches with their severe effects. Undoubtedly along the way we shall learn a great deal more about the mechanisms of lubrication and wear in general.

In particular this is of interest in the case of lubricated contact where predictability is the poorest. We should find out when flat sliding surfaces (such as the sides of pistons) develop the thermal asperity, or else discover the mechanisms which prevent this.

Although journal bearings have been investigated as to radial clearance loss due to thermal deformation, this writer knows of no investigation of ellipticity or multilobar deformation patterns which might be thermally excited and could lead to catastrophic failure.

In other bearing configurations such as tilted pads one can visualize a thermal asperity forming at high speed. Again it should be determined whether or not this actually happens in practice.

It is this writer's opinion that even the minimum film thickness criterion of bearing failure should be re-investigated with a thermo-elastic analysis of the thin film region to determine the possibility of a load-concentrating temperature rise and thermal breakdown of the film.

It is clear that thermoelastic effects distort gas bearings, but as far as this writer can determine no stability analysis has been made of such systems. Furthermore although ball bearings have been shown to be possibly subject to thermoelastic seizure processes, the study of thermoelastic effects in catastrophic failures is in the most primitive stage of development and should be extended.

Thermoelastic deformation of supposedly non-contacting jet engine shaft seals appears on initial investigation to have a strong likelihood of occurring, and could lead to thermal asperities on the seal face or concentration of load on two or more pads with attendant increased leakage, and possibly failure. The mating rings may also deform significantly.

The high-speed rubs of turbomachines in the form of blade tip contact and labyrinth seal contact are already under investigation. Further investigation of thermoelastic deformations brought about by such contact may help to provide guides to improve designs.

Granting that thermal asperities do appear in important applications such as seals, it is essential to understand the thermoplastic behavior of the patches and to estimate the stresses, metal working, and fatigue mechanisms in such contacts.

In conventional face seals, where the effects are most easily demonstrated, the complex interactions of tilt, nutation and other motions with those associated with thermal deformation can only be conjectured, yet experiments which have been made suggest that these interactions are complex and closely intercoupled. Stress analysis for the thermoelastic regime for hard materials should be undertaken and the conditions for stress cracking (heat checking) should be defined.

Turning now to physically different problems, Marshall (51) has reported contact patch formation in high-current electrical brushes. It appears that current as well as friction can produce the deformed state, and the combined effects of these should be studied to determine their influence upon wear as well as the limits of operation of brushes.

Although no publications have appeared on the subject, this writer has been told that thermoelastic instability, involving roller deformation, sets the limit for speed of rolling of thin sheet. Rollers may be undercut to compensate for "barreling out" as a result of heating. However, at some limiting speed this is no longer effective in assuring uniform sheet.

Recently Quinn has given evidence for the thermoelastic oscillation of asperities in a concentrated, rider-on-drum contact, and has provided estimates of the number of asperities in contact at one time. On the basis of earlier theories thermal augmentation of asperities would not have been expected in the small dimensions involved; but the evidence is compelling.

Summarizing, it is this writer's opinion that thermal deformation and the formation of the thermal asperity are not isolated phenomena to be encountered in special circumstances. They appear and sometimes become predominant in quite ordinary applications. Indeed, it is believed that neglect of accounting for them has needlessly complicated the interpretation of many friction and wear observations. Increased understanding has led to improved tools to use in studying these effects. To apply these to some of the obvious problem areas listed above should provide in some cases improved physical understanding, in others design guides and in a few a clearing of the air by showing that such effects are under control.

TABLE 1

Properties of Representative Materials

	Aluminum	Cast Iron	Silicon Carbide	Graphite	Glass
E(megapascals)	6.8×10^5	1.2×10^5	8×10^5	0.68×10^8	8×10^5
$\alpha(1/^{o}C)$	17×10^{-6}	11×10^{-6}	4.7×10^{-6}	4.7×10^{-6}	5.4×10^{-6}
$K(N/sec\ ^{o}C)$	227	50.4	18	13.5	0.9
$k(cm^2/sec)$	83×10^{-2}	11×10^{-2}	6×10^{-2}	7.9×10^{-2}	0.31×10^{-2}

TABLE 2

Drum	Blade	Friction Coef, μ	V*(m/sec)	measured w	calculated w_c
$A\ell_2O_3$	$A\ell$	0.38	7.11	14×10^{-10}	13×10^{-10}
"	Brass	0.16	8.13	9×10^{-10}	1.7×10^{-10}
"	Steel	0.26	5.08	2.5×10^{-10}	1.1×10^{-10}
Steel	Steel	0.25	12.20	13×10^{-10}	0.5×10^{-10}
		Lubricated			
Steel	Alum. Graphite	0.026	32	0.26×10^{-10}	<0
"	Alloy Steel	0.051	23.9	0.54×10^{-10}	1×10^{-13}

Figure Captions

(1) Cross-section of apparatus where ring contact occurs between an inverted cup and a glass plate. Probe A reads surface profile of the cup, while B indicates overall rising or bouncing.

(2) In these oscilloscope traces of probe outputs, upper trace is probe B and lower is probe A. As speed is increased moving waves grow in cup face and at 13 m/sec there is a sudden formation of a thermal asperity. Slight bouncing is indicated as this goes over the probe indentation. Upon lowering speed (plate 6) the thermal asperity disappears except for a tiny wear scar.

(3) Critical disturbance velocity in the less conductive body for a range of friction coefficients and for aluminum with hypothetical reduced conductivity sliding on aluminum.

(4) Effect of thin glass films on $A\ell$ as to critical disturbance velocity.

(5) Effect of wear and heat transfer on stability of a scraper, as ζ increases heat transfer increases to surface scraper slides against.

(6) Radial hairline cracks across the contact band (light area) on the metal face of a seal, after running against a carbon ring at high peripheral speed.

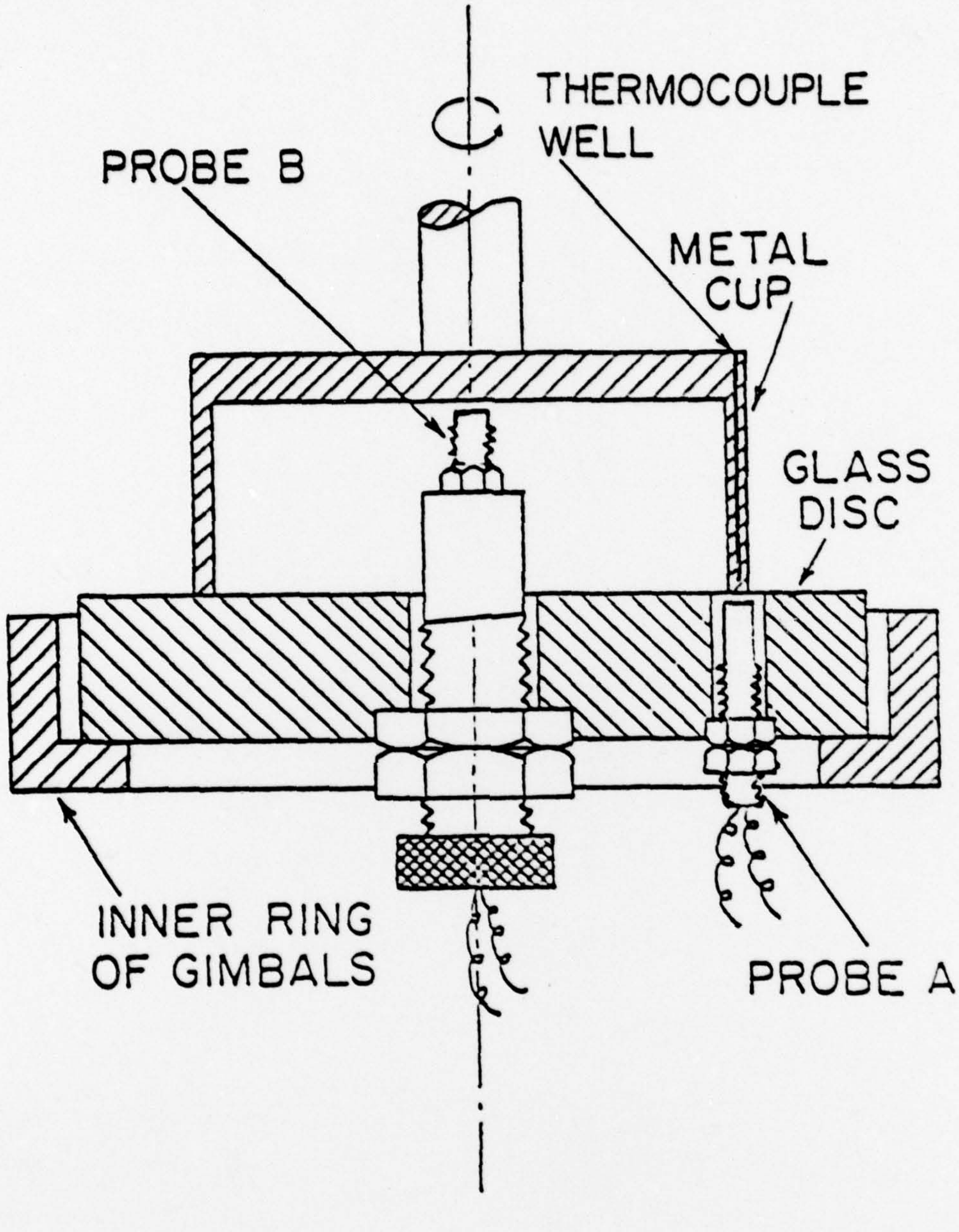

THERMOCOUPLE
WELL
PROBE B
METAL
CUP
GLASS
DISC
INNER RING
OF GIMBALS
PROBE A

FIG. 1

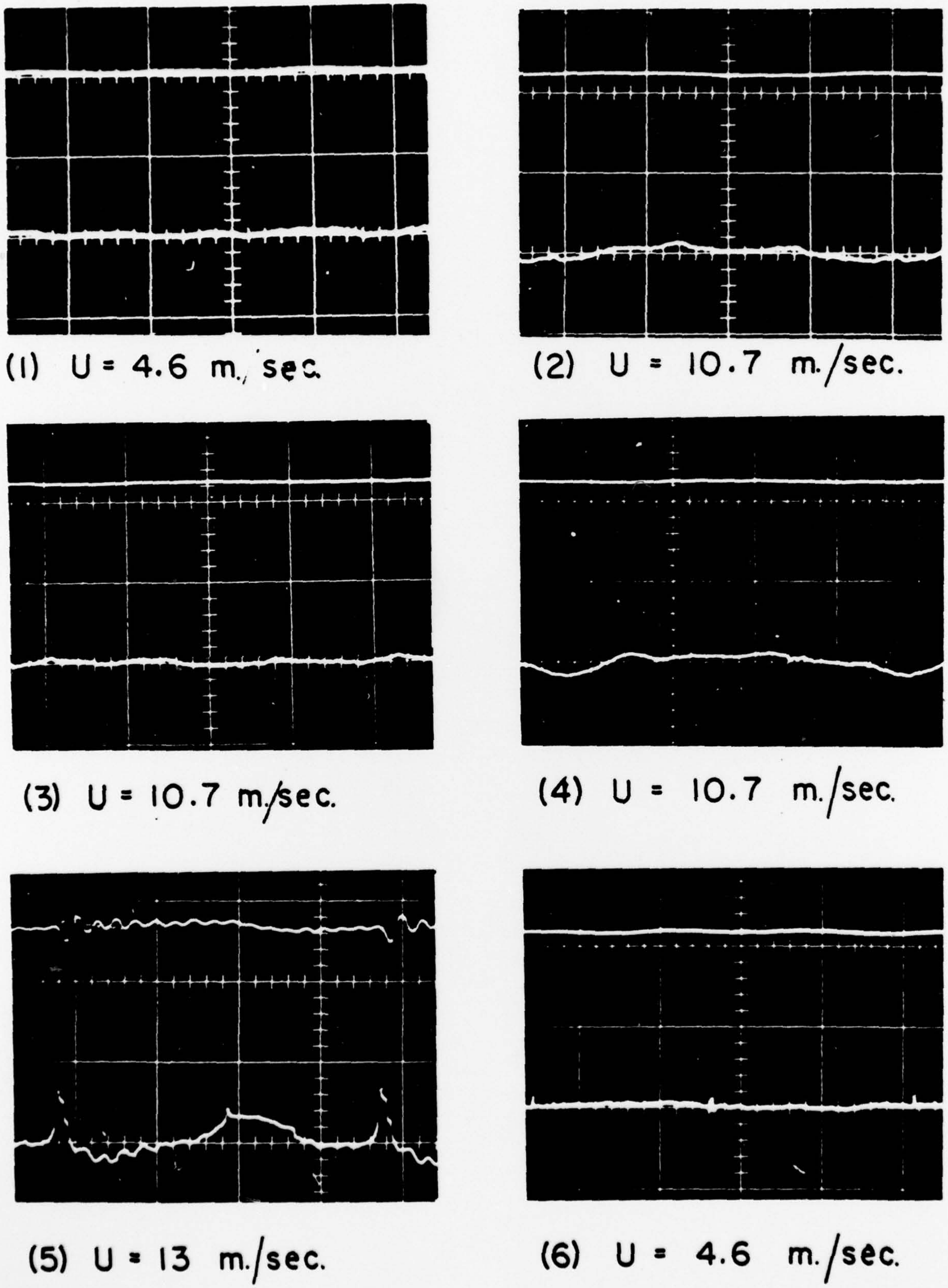
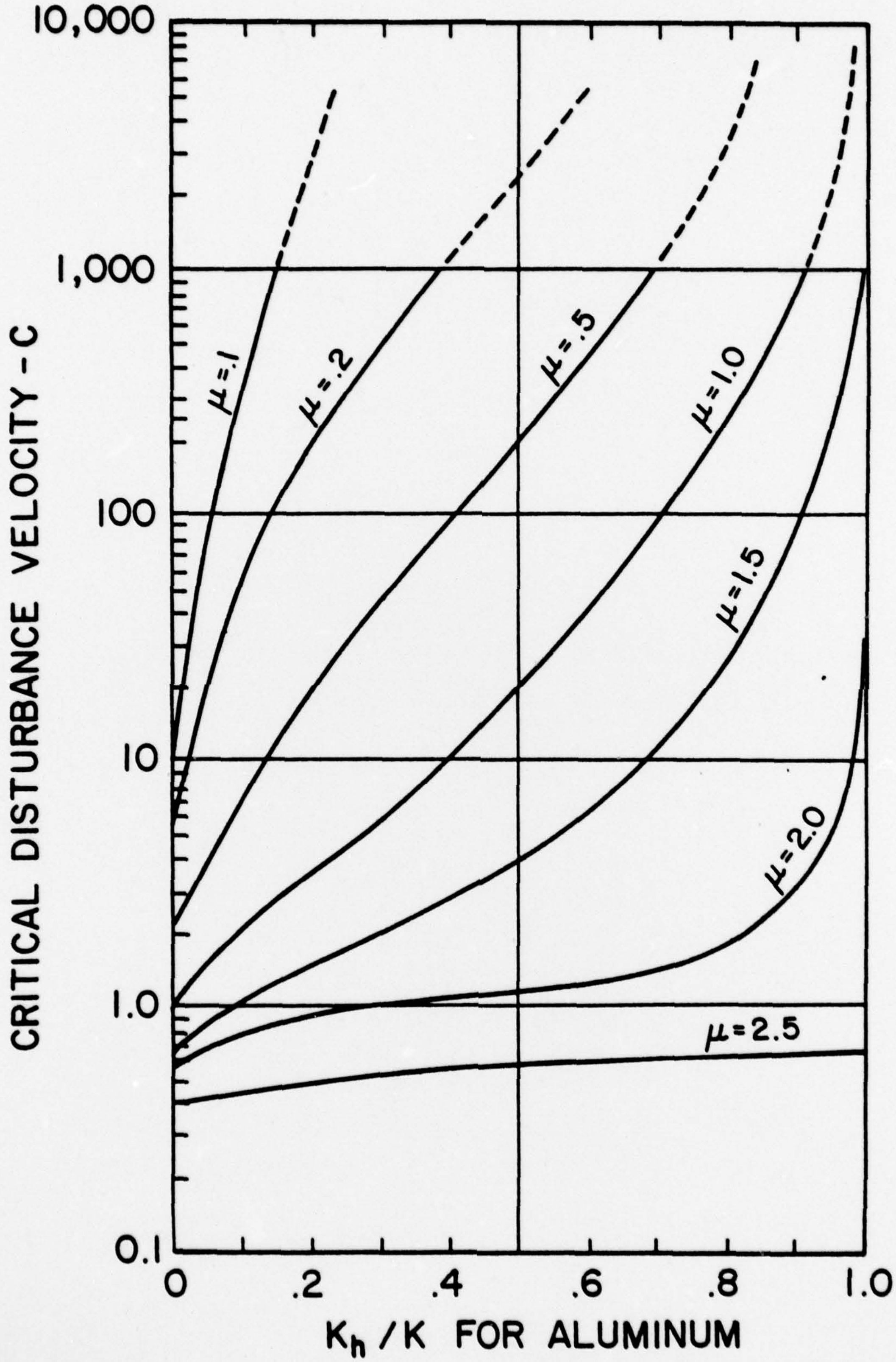
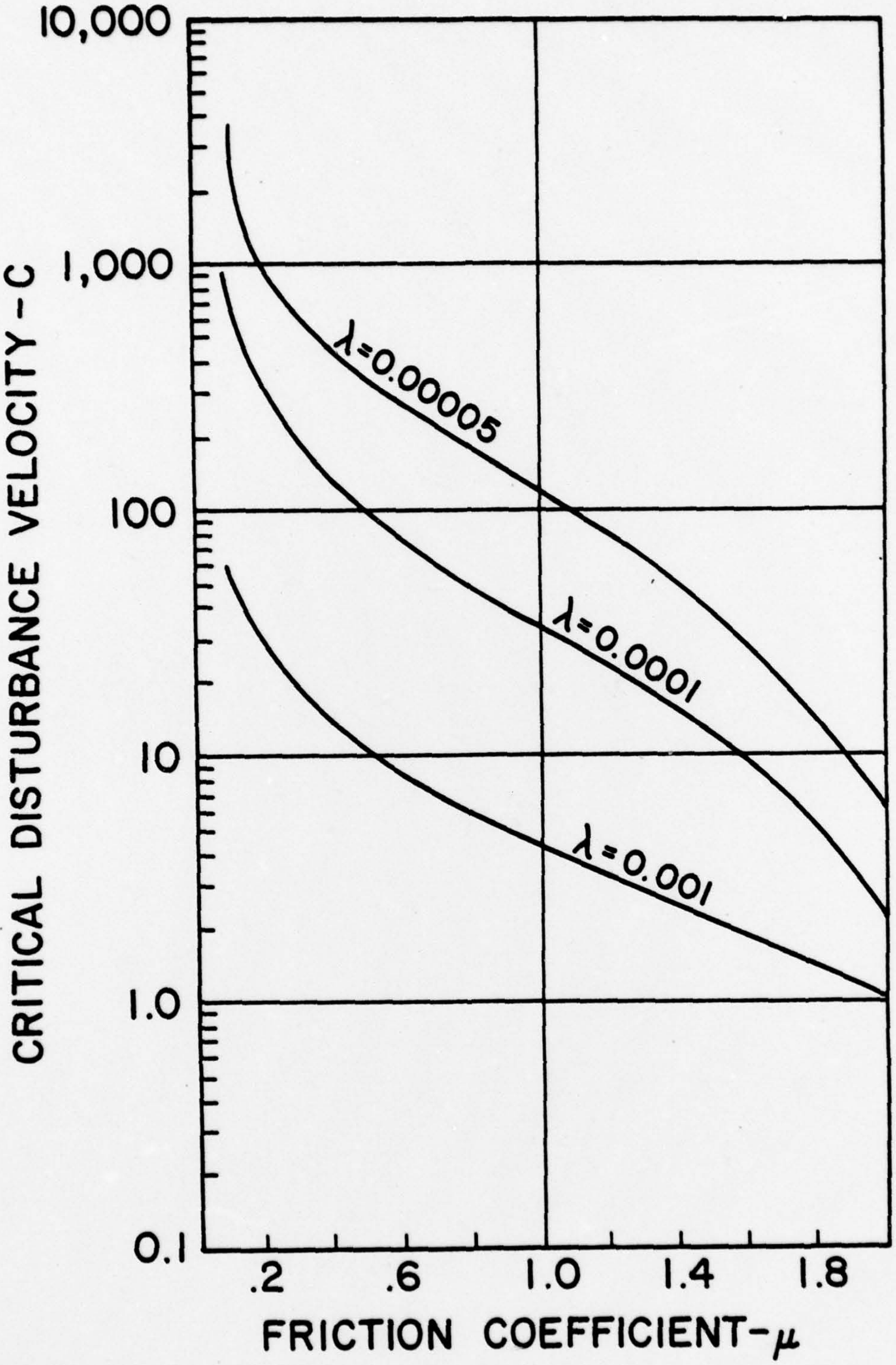

(1) U = 4.6 m./sec.
(2) U = 10.7 m./sec.
(3) U = 10.7 m./sec.
(4) U = 10.7 m./sec.
(5) U = 13 m./sec.
(6) U = 4.6 m./sec.

FIG. 2



10,000
1,000
100
10
1.0
0.1
CRITICAL DISTURBANCE VELOCITY - C
μ=.1
μ=.2
μ=.5
μ=1.0
μ=1.5
μ=2.0
μ=2.5
0
.2
.4
.6
.8
1.0
K_h / K FOR ALUMINUM

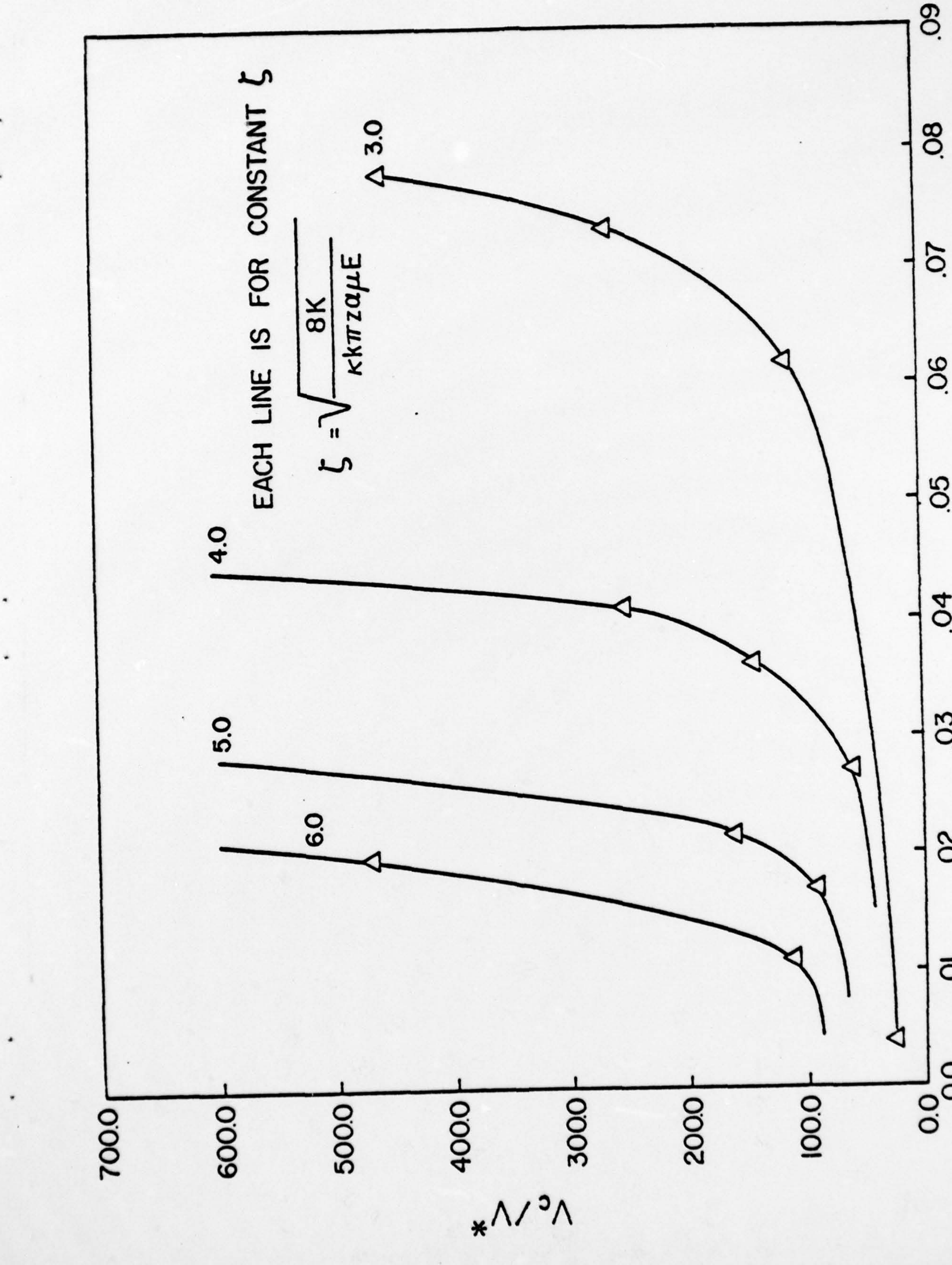
FIG. 3



10,000
1,000
100
10
1.0
0.1
CRITICAL DISTURBANCE VELOCITY - C
λ=0.00005
λ=0.0001
λ=0.001
.2
.6
1.0
1.4
1.8
FRICTION COEFFICIENT-μ

FIG. 4

FIG. 5



EACH LINE IS FOR CONSTANT ζ
$\zeta = \sqrt{\dfrac{8K}{\kappa k \pi z a \mu E}}$
3.0
4.0
5.0
6.0
V_c/V^*
700.0
600.0
500.0
400.0
300.0
200.0
100.0
0.0
0.0
.01
.02
.03
.04
.05
.06
.07
.08
.09
W/W*

Fig. 6

List of Symbols

b	parameter in heat equation
c	speed of movement of contact patch along surface
E	Young's modulus
E^*	modified modulus for two body contact: $1/(1/E_1 + 1/E_2)$
h	axial length of sealing ring
$\bar{h}$	film thickness in seal
k	thermal diffusivity
K	thermal conductivity
ℓ	half-length of contact patch
p	pressure
q	heat flow through unit area of surface
Q	total heat flow through contact patch
R	radius of curvature of surface
t	time
T	temperature
V	sliding speed
V^*	critical sliding speed
w	wear coefficient
w^*	critical wear coefficient
x	coordinate along direction of sliding
y	coordinate normal to surface
Pe	Peclet number
Pe_ℓ	Peclet number of contact patch

α	coefficient of thermal expansion
ζ	heat transfer parameter
η	viscosity
κ	wave number
λ	wave length
μ	friction coefficient
ξ	dummy variable
$(\hat{\ })$	implies amplitude of wave
$(\overline{\ })$	implies mean value

REFERENCES

1. R. Holm, Wiss. Veroeff. a.d. Siemens Konzern, Berlin, 7, 2 (1929) 217-271.

2. F. P. Bowden and D. Tabor, The Friction and Lubrication of Solids, Oxford University Press (1950).

3. M.E. Merchant, J. App. Phys., 11 (1940) 230.

4. M. Belidor, Architecture Hydraulique, C. A. Jambert, Paris (1737).

5. C. Coulomb, Memoires de Mathématique et de Physique, Académic des Sciences, 10 (1785).

6. H. Blok, I. Mech E. Proc., Lubrication Discussion, Group IV, 2 (1937) 26-39.

7. R. Holm, J. App. Phys., 19 (1948) 361-366.

8. J. F. Archard, Wear, 2, 6 (1959) 438-455.

9. F. F. Ling and T. E. Simkins, Trans ASME, Ser. D, 85 (1963) 481-487.

10. F. F. Ling, Surface Mechanics, John Wiley and Sons, New York (1973) 125-132.

11. V. C. Mow and H. S. Cheng, ZAMM, 18 (1967) 500-507.

12. M. V. Korovchinsky, Trans ASME, Ser. D, 87 (1965) 811-817.

13. A. Nica, Trans ASME, Ser. D, 87 (1965) 781-784.

14. R. A. Burton, Wear, 8 (1965) 157-172.

15. Y. C. Hsu, Trans ASME, Ser. F, 89 (1967).

16. H. E. Staph, ASLE Trans, 10 (1967) 408-417.

17. L. B. Sibley and C. M. Allen, ASME Paper 61-Lubs-15.

18. J. R. Barber, Dissertation, St. John's College, Cambridge, England (1968).

19. J. R. Barber, Proc. Reg. Soc., A, 312 (1969) 381-391.

20. J. R. Barber, Wear, 10 (1967) 155.

21. T. A. Dow and R. A. Burton, Wear, 19 (1972) 315-328.

22. T. A. Dow and R. A. Burton, Trans ASME, Ser. F, 95 (1973) 71-75.

23. T. A. Dow, Dissertation, Northwestern University (1972).

24. T. A. Dow and R. D. Stockwell, Trans ASME, Ser. F, 261 (1977) 359-364.

25. F. E. Kennedy and F. F. Ling, Trans ASME, Ser. F, 96 (1974) 496-507.

26. J. J. Santini and F. A. Kennedy, Lubr. Eng., 31 (1975) 402-409, 413-417.

27. R. A. Burton, V. Nerlikar, and S. R. Kilaparti, Wear, 24 (1973) 169-188.

28. R. A. Burton, Wear, 24 (1973) 189-198.

29. R. A. Burton and V. Nerlikar, Trans ASME, Ser. F, 97 (1975) 539-545.

30. A. O. Lebeck, Trans ASME, Ser. F, 98 (1976) 277-285.

31. R. A. Burton, S. R. Kilaparti and V. Nerlikar, Wear, 24 (1973) 199-206.

32. R. Kilaparti, ASLE Trans, 20 (1977) 64-70.

33. R. Kilaparti, Trans ASME, Ser. F, 100 (1978) 65-69.

34. F. F. Ling and V. C. Mow, Trans ASME, 87 (1966) 814-816.

35. B. N. Banerjee and R. A. Burton, Trans ASME, Ser. F, 98 (1976) 157-166.

36. B. N. Banerjee and R. A. Burton, Nature, 261 (1976) 399-400.

37. B. N. Banerjee, Dissertation, Northwestern University (1977).

38. E. J. Hahn and C. F. Kettleborough, Trans ASME, Ser. F, 90 (1968) 233-239.

39. K. Rockwell, C. Ettles and M. Stokes, I. Mech. E. Brighton, 184, Part 3 L, Paper 12, (1970) 82-92.

40. S. Tanaguchi and C. Ettles, ASLE Trans, 18 (1975) 299-305.

41 S. R. Heckmann and R. A. Burton, Trans ASME, Ser. F, 99 (1977) 247-253.

42. S. R. Heckmann and R. A. Burton, Trans ASME, Ser.F, 100 (1978) 136-141.

43. R. A. Burton, The Mechanics of Contact Between Deformable Bodies, Delft University Press, Delft (1975) 191-205.

44. C. Wangkrajong, M.S. Thesis, Mechanical Engineering Dept., Northwestern University, Evanston, IL 60201.

45. R. A. Burton, S. R. Kilaparti and S. R. Heckmann, Trans ASME, Ser. A, 98 (1976) 435-440.

46. H. R. Wetenkamp and R. M. Kipp, ASME Paper 75-RT-2.

47. J. F. Archard, J. App. Phys, 24 (1953) 981.

48. J. Dundurs, Mechanics Research Communications, Pergamon Press, 1 (1974) 121-124.

49. J. Dundurs and C. Panek, Int. J. of Heat and Mass Transfer, 19 (1976) 731-736.

50. R. Kilaparti and R. Burton, ASME Trans, Ser. F, 98 (1976) 596-601.

51. R. A. Marshall, Wear, 37 (1976) 233-240.

52. T. F. J. Quinn, ASLE Trans., 21, (1978) 78-86.